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CHAPTER 3
Spillway Gate Operating Equipment

3.1 Tainter Gate.

3.1.1 General. The tainter gate is considered the most economical, and usually the most suitable, type of gate for controlled spillways because of its simplicity, light weight, and low hoist-capacity requirements. The principle elements of a tainter gate structure are the skinplate assembly, the members supporting the skinplate assembly, the end frames, the trunnions, the anchorages and the hoisting machinery. Tainter gate design information is provided in EM 1110-2-2702. This sub-chapter presents information about wire rope electric motor driven, direct connected hydraulic cylinder, round link chain pocket wheel, and round link chain grooved drum hoisting machinery that have been successfully used in the past. Whenever possible and practicable the designer should consider incorporating redundancy into the gate operating system.

3.1.2 Wire Rope Electric Motor Hoist. This type of machinery usually consists of two similar but opposite-hand hoist units mounted on piers and arranged to lift each end of the gate. The drive unit usually consists of electric motor, worm reducer, DC magnet brake, parallel shaft speed reducer, open gearing and rope drum. The driven side contains only the enclosed speed reducer, open gearing and rope drum. The two units are kept in synchronism by a cross shaft carried in bearings that are supported from a bridge connecting the two piers. A general arrangement of a wire rope electric hoist is shown on Plates B-71, B-72 and B-73. It should be noted that the hoist has been so located on the pier that the wire ropes will contact the skinplate for essentially the full height of the gate. This is done to prevent floating debris from becoming lodged between the gate face and the wire rope and possibly damaging the rope. Also, the input shaft of the driven side parallel shaft speed reducer must be of special design suited for the total torque required to lift the gate. Wire rope hoists are very common for new construction and when rehabilitating or replacing existing spillway operating machinery.

3.1.2.1 Design Capacity. The design capacity for an electric motor driven hoist should be based on the maximum load at normal speed. Loads on the component parts should include the above loads and the applicable friction losses in the reducers bearings, gear trains, and rope. Experience has shown that allowances for friction losses in the hoist components should be liberal. Losses in speed reducers when operating at low speed are greater than those listed in manufacturers literature and are further increased by high viscosity of lubricants in low ambient temperatures. Heaters have been provided in speed reducers where low temperature operation is required. When used, heater elements should be selected with low watt density 1.5-1.8 watt/cm^2 (64-77 watts/in^2) to prevent charring the oil. More recently, synthetic oils with high viscosity index have been found suitable for use in speed reducers operating at low temperature. When synthetic oils are considered, proper seal selection for the speed reducer is essential due to the aggressive nature of these oils. However, the use of synthetic oils reduces maintenance and energy cost associated with heaters.

3.1.2.2 Component Parts. Component parts of wire rope electric hoists are designed for a factor of safety of 5 based on normal loading and, in addition, each part is designed for a unit stress not in excess of 75 percent of the yield point of the material under loads resulting from the maximum torque of the motor selected. Both normal loads and loads resulting from the maximum torque of the motor (usually 280 percent of full load torque) should be considered as equally divided between the two drives of a hoist. Shock, impact, and wear factors are considered negligible and may be disregarded.

3.1.3 Direct Connected Hydraulic Cylinder. This type of machinery usually consists of two hydraulic cylinders, one mounted on each pier and arranged to lift each end of the gate. The cylinders are trunnion type and mounted in cardan rings which are supported by hoist frames cantilevered over the side of the pier. The piston rod is connected to the gate through a spherical bearing. A general arrangement of a direct connected hydraulic cylinder type tainter gate hoist is shown on Plate B-74. Details of the mounting arrangement are shown on Plate B-75. Individual hydraulic power units are usually mounted in rooms at the top of each pier although an arrangement with a single power unit is possible. As much valving as possible is mounted in manifolds connected directly to the cylinder ports. This includes a pilot operated check valve on the rod end port used to hold the gate in a raised position. This arrangement minimizes interconnecting piping and the potential for leakage or failure. Plate B-76 shows a typical hydraulic schematic diagram. See EM 1110-2-2702 for loading conditions associated with direct connected hydraulic cylinder operated tainter gates. Direct connected hydraulic cylinders are gaining popularity for spillway gate operating machinery for new gated spillway construction. Converting an existing wire rope or chain gate operated hoist to hydraulic cylinder will be both difficult and costly due primarily to design modifications to the piers that support the cylinders.

3.1.3.1 Piston Rod. The piston rod is usually connected through a spherical bearing to a lower framing member on the gate necessitating that the gate arms be skewed to the pier rather than parallel to the pier. It is also possible to connect the piston rod directly to the top of the side arm which could then be parallel to the pier.

3.1.3.2 Cylinder Synchronism. The two cylinders are kept in synchronism by the hydraulic controls. Position indicators mounted internal to each cylinder provide a signal, relative to cylinder stroke, to the control system. This system generates an error signal which is used to control a small proportional valve. This valve is used to bleed oil from the rod side of the lead cylinder when raising and from the rod side of the lag cylinder when lowering. A typical schematic diagram with description of components is shown on Plate B-76. For small gates or gates that are infrequently operated and then returned to the sill, such as on flood control spillways, a simpler system utilizing a flow divider may provide sufficient synchronization.

3.1.3.3 Design Capacity. The design capacity of the hydraulic cylinder should be a minimum of 125 percent of the maximum load. This resulting design capacity should be used to establish the system operating pressure, cylinder bore and piston rod diameter. All components of the hoisting system should be designed in accordance with the provisions of UFGS-15010A, Hydraulic Power Systems for Civil Works Structures. Appropriate consideration should be given to the selection of the fluid to be used in the system in regard to anticipated operating temperature. The spherical bearing used at the gate connection should be constructed of corrosion resisting materials and be of the self lubricating type.

3.1.4 Round Link Chain Hoists. Round link chain hoists presents both pocket wheel and grooved drum lifting mechanisms. These type of hoists have been used primarily to replace existing roller chain hoists.

3.1.4.1 Pocket Wheel. The pocket wheel, shown on Plates B-77 & B-78, is a universally applied lifting mechanism to handle and hold a round link chain to the limit of the chain's breaking strength. A pocket wheel is designed to properly load the chain in tension and bearing without inducing the bending loads predominant in a grooved drum. The wheel may be either a ring forging of alloy steel or a weldment.

3.1.4.1.1 Design Standard. A standard specification for a ring forging is provided below:

Material	ASTM A290, Class K
Tensile strength, minimum	1,170 Mpa (170,000 psi)
Yield strength, minimum	1,000 Mpa (145,000 psi)
Brinnel Hardness range	340 to 400

The standards for the design of a pocket wheel are derived indirectly from the dimensions included in DIN 22252, Part 1 (High-tensile Round-link Steel Chains for Mining; Testing). This standard covers the dimensions and tolerances for chain that is compatible with pocket wheels. Preliminary design calculations for pocket wheels using a specific chain size are necessary in order to determine that the unit size is compatible with any physical space limitations imposed by the gate machinery location. An additional auxiliary item required for a pocket wheel mechanism is a chain locker. The size of a chain locker should be such that it adequately contains the slack length of chain when the gate is in the fully raised position. Chain locker volume should be a minimum of the product of the diameter of the chain in inches squared, times the length of the chain in fathoms, times 0.85. A sample calculation is included in the Appendix C showing various dimensions of chain lockers required for a 14-meter (46-foot) length chain 38 mm (1.5 in.) in diameter.

3.1.4.1.2 Availability. While no manufacturer will have a standard off-the-shelf product that will fit a given application exactly, the technology to build a pocket wheel to a given design criteria is available by many manufacturers. The pocket wheel has been successfully installed at Locks and Dams on the Upper Mississippi and Illinois Rivers.

3.1.4.1.3 Assembly Test. After installation is completed, an assembly lift test should be required as part of the contractor's responsibility for the gate lifting machinery. These tests should include not only a design load test, but an overload test that proves that the maximum specified motor torque will not deform the chain or allow the chain to slip on the pocket wheel.

3.1.4.2 Grooved Drum. Another type of round link chain lifting mechanism is a cylindrical grooved drum. This design includes a cast or fabricated cylinder with a helical groove that is either cast or turned into the surface. The groove is designed to accept every other chain link and must be sized large enough to wind the entire length of chain around the drum in a single row. One advantage of a grooved drum is that it requires no chain locker since it stores the chain on the drum similar to a wire rope drum. The diameter of the grooved drum should not be less than 25 or 30 times the diameter of the bar used for the chain links. For applications where there is no additional room for chain storage, the use of a grooved drum may be indicated. An added advantage of this type of drum is that it is able to accept a deformed link without becoming jammed. The main disadvantage of the grooved drum is the manner in which it loads the chain links. Each link is loaded in both tension and bending. This loading situation puts undue stresses on the links, especially since chain links are not meant to be loaded in bending. A sample stress calculation is included in Appendix C. It shows a 38 mm (1.5 in.) chain being loaded on a 1.0 meter (41.49 in.) diameter drum. The results of the calculation show that, even under a normal tensile load, the chain is loaded at or near its yield point. Chain is typically known by its breaking strength and a proof load is normally specified. Actual stress levels in a chain under loading are determined by an involved analysis dependent on criteria such as chain geometry, hardness, material properties, etc.

3.1.4.3 Round Link Chain. The selection of a chain handling device depends entirely on the type of chain to be used. There are many types of chain commercially available today for various applications of lifting service. The links of the round link chain are not actually round, but have round ends and approximately parallel sides. The calibrated links are designed specifically to be used with a pocket wheel that drives the chain properly, loads each link in tension and bearing, and eliminates the bending stresses in the links that occur when a grooved drum is used. This type of chain is widely applied to both low speed and high speed lifting and is both abrasion and corrosion resistant.

3.1.4.3.1 Material. Round link chain used in chain hoists, shown on Plate B-79, is made from an alloy steel. Although the materials and heat treatment may vary among manufacturers, AISI 8620 is a common material for this chain. This material is heat treated to a tensile strength of approximately 965 Mpa (140,000 psi). The hardness of this chain from different manufacturers may also vary, but a figure of 300 BHN is considered average. The higher hardness of this material provides improved wear qualities over low alloy chain. However, low alloy chain of equal breaking strength has a greater energy absorbing capacity, and, therefore, greater shock load capacity, than high alloy chain.

3.1.4.3.2 Compatibility with Existing Material. The type of chain that may be used for gate hoisting should be compatible with the existing gate and hoist component materials to prevent undue wear and abrasion. Hoisting chains mainly contact the wear bars or wear plates on the surface of the gate. The wear bars on the gates should be sized specifically for the chains that will be installed. Wear bars and plates should be compatible in size and hardness with the lifting chain selected. Galvanic corrosion due to dissimilar metals should also be considered during the design process.

3.1.4.3.3 Tolerances. Manufacturing standards for calibrated round link chain require that each length of chain meet certain tolerances with regard to link size and breaking strength. High alloy hoist chain is manufactured to length and width plus or minus 0.51 to 0.76 mm (.02 to .03 in.) These tolerances are an international standard so that all chain, regardless of manufacturer, will be suitable for the intended use. The DIN standards for strength testing of this chain are very rigorous and include tensile, bending, and shock tests.

3.1.4.3.4 Abrasion. For chain to be suitable for dam gate lifting service, it must be resistant to abrasion caused by silt trapped in the submerged links. Chain used for dam gate lifting will probably never be washed or cleaned since it would be difficult and impractical to do so. High alloy conveyor chain must be specifically to be very abrasion resistant.

3.1.4.3.5 Shock. To resist shock loads, a material must be strong and be able to absorb the energy imparted to it by the shock load. When carbon steel is alloyed and heat treated to increase strength, its energy absorbing capacity doesn't increase proportionately. Thus, for equal breaking strength, lower alloy material will normally be more resistant to shock loading than higher alloy material. Round link alloy chain is specifically tested for shock loads by the manufacturer.

3.1.4.3.6 Distortion. To be compatible with a hoist chain pocket wheel, the chain must be capable of being loaded without being significantly distorted. If the chain were to become distorted, which usually amounts to the link becoming longer and narrower, the links would no longer fit the lifting device pockets. Round link chain resists distortion since the sides of the link are designed to remain parallel. If these links distort, they tend to elongate, but they can only do so after they have exceeded the elastic limit of the material. The design criteria for lifting chain requires that the minimum breaking strength of the chain shall be no less than 5 times the design load, and that the lifting machinery shall in no case impart to the chain a load that will exceed 75 percent of the yield strength of the steel in the chain. When a chain is selected within these design limits, link distortion will not be a factor.

3.1.4.3.7 Corrosion. The type of chain discussed in this chapter of the EM should normally be protected against corrosion. Hoist chain is available with a variety of special corrosion coatings such as special paints or hot galvanizing. It should be noted that hot galvanizing should not be used for chain in this type of application unless the reduced strength due to reheating is taken into account. While selecting a corrosion coating is not part of the scope of this manual, it should be noted that the existing chains on the prototype pocket wheels at Lock and Dam 20 on the Mississippi River have worked very well for a period simulating 50 years of operation, with no corrosion coating at all.

3.1.4.3.8 Replacement. Replacement of round link chain described in this manual should not be a problem in the foreseeable future. This type of chain is widely used in the U.S. and in foreign countries.

3.1.4.3.9 Chain Costs. The actual competitive costs for chain can be accurately determined only after bidding. However, for comparison purposes and cost estimates, cost figures for chain in the round link hoist category can be obtained from various manufacturers. Chains should meet the strength and durability requirements for gate lifting service at Civil Works Projects. Differences in cost could influence the chain selection criteria in a project similar to the one Lock and Dam 20 (Mississippi River), which required almost one mile of chain to power all the gates.

3.1.5 Engineering Steel Chain.

3.1.5.1 Introduction. In addition to round link chain, engineering steel chain should also be considered as a replacement for existing roller chain. Roller chains (using pins, rollers, and sidebars) have been, for many years, a source of operation, maintenance and environmental problems at gated spillways owned and operated by the Corps. Past roller chain design is difficult to lubricate, causing bearing surfaces to corrode and bind which prevents smooth operation of the chain over the sprocket. As a result, spillway gates could not be operated, chains failed, and gates dropped, creating both a dam safety problem and a safety hazard for operating personnel. The chain design that's described herein has solved the problems associated with past roller chain design for both tainter and roller gates in both the Huntington and Saint Paul Districts. It has been in use by these districts since 1997 with no problems reported. The following topics are presented: material selection, corrosion prevention, first cost, and life cycle cost. An engineering analysis of this type of chain is presented and maintenance issues examined.

3.1.5.2 Terminology. It is important to differentiate how a lifting chain for a tainter gate is different then a bicycle chain beyond the obvious size and strength differences. There are several chain standards and a chain manufacturer's association that classifies various chain types.

The chain industry, chain manufacturers, and American Chain Association (ACA) make a distinction between Roller Chain and Engineering Steel Chain. In general, Roller Chain is used for power transmission between sprockets at moderate to high speeds. The chain speed, sprocket design, and kinematics between the sprocket and chain are crucial. Roller Chain is manufactured per ANSI/ASME B29.1 (see standards paragraph below). The tension members between pins (side plates) are called link plates. This type of chain is generally produced in large quantities. The size and strength ratings are relatively low.

Engineering Steel Chain is intended for a wider variety of applications including materials handling, conveying, and other industrial uses. The Engineering Steel Chain is usually manufactured in smaller quantities, has greater strength, more corrosion resistance, greater shock resistance, and designed to be used in severe environments. The chain is manufactured per several standards including ANSI/ASME B29.10 and B29.15. The side plates are called sidebars. The pin-bushing area is referred to as the chain joint. The sidebars establish the chain pitch (see Figure 3-1).

The ACA defines tension linkage chain as a chain application where the main function is to move a load slowly, intermittently through a short distance, or to hold a load. These types of

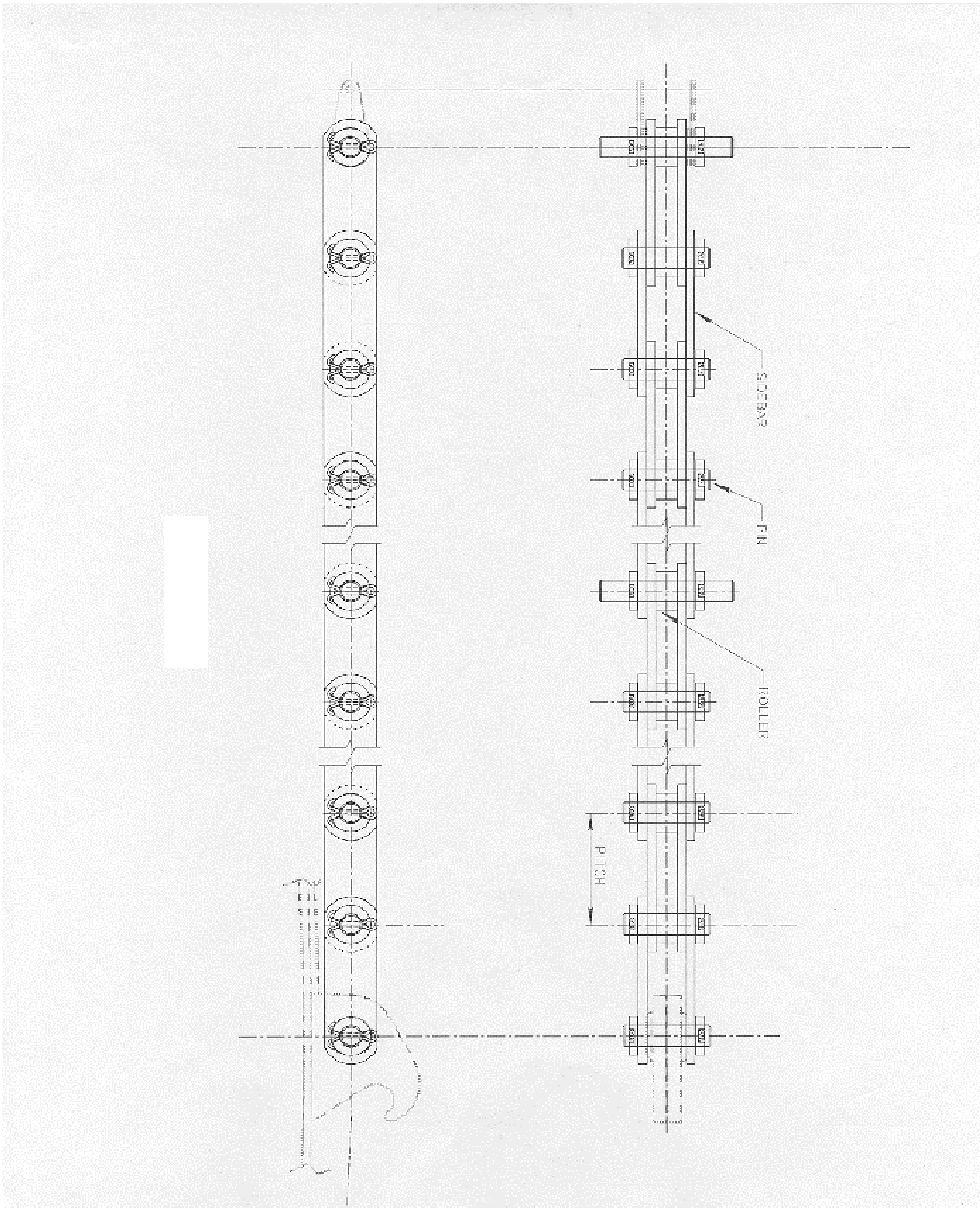


Figure 3-1. Chain Assembly

chains are used for hoisting, supporting counterweights, etc. The function of a tension linkage chain is to transmit a moving force using chain tension hence the nomenclature.

Lifting chain for roller and tainter gates thus falls under the category of a tension linkage, Engineering Steel Chain. The following definitions are provided:

- Pitch is the distance between the centers of adjacent chain joint members or center-to-center distance between adjacent pins,
- Sidebars are tension members connecting the chain joints, and
- Pins connect one link section to another. Pins are the shear members between the inner and outer sidebars.

3.1.5.3 Standards. The American National Standards Institute (ANSI) and American Society of Mechanical Engineers (ASME) both publish standards for chain as stated above. These standards are written in English units although the latest editions of these standards have the metric equivalent (for reference only). In fact, the basic design of the chain in ANSI B29.1 is written in English units. The roller diameter is defined as 5/8 times the pitch. There are manufacturers, however, that make a true metric chain. Also, the European DIN (German) Standards and the International Standards Organization (ISO) categorize metric chain.

Many of the original tainter gate chain designs (from the 1930's) for the Upper Mississippi River projects used offset side bar roller chain as opposed to straight side bar roller chain. The primary benefit of offset side bar type of chain is that the links are all identical. Offset sidebar chain can be used in odd or even number pitches. The primary advantage of using straight sidebar chain is that the chain is easier to manufacture and for a given sidebar plate thickness, the straight sidebars will have more strength. Straight sidebar chain consists of inside and outside links and sections of this chain type must be used in even number of pitches (lengths). This chain can also be constructed without rollers. However, in gate lifting applications, the rollers are necessary for reducing friction as the chain is going over the sprocket.

It should also be noted that the majority of the ANSI/ASME standards concern chain used in power transmission rather then lifting applications. However, this difference is generally irrelevant. The loading on the chain is basically the same and in both applications the chain is going over a sprocket. The biggest difference between power transmission and lifting application is likely to be the speed. In lifting applications, the chain travel will be extremely slow.

ANSI/ASME B29.1M (1993), Precision Power Transmission Roller Chains, Attachments, and Sprockets list a series of standard roller chain. However, this standard only classifies chain up to a pitch of 3.0 inches. ANSI B29.1 assigns standard number designations to chain based on pitch, chain width,
and roller diameter. The chain sizes given in B29.1 are generally inadequate for a majority of tainter gate and roller gate lifting applications. The primary benefit of this standard is that any chain manufactured according to it will fit over any corresponding sprocket manufactured to the standard. The chain of one manufacturer will replace the chain of another manufacturer.

ANSI/ASME B29.10M (1994), Heavy Duty Offset Sidebar Power Transmission Roller Chains and Sprocket Teeth, only standardizes offset sidebar type of chain. This standard is an Engineering Steel Chain standard and includes chain with a pitch up to 7 inches (177.8 mm) and

a minimum ultimate strength of 425,000 pounds (1890 kn).

3.1.5.4 Material Selection. Material selection is likely the single most important feature of the lifting chain design. The type of material used for the chain will impact the strength, corrosion resistance, and overall life of the chain. Proper material selection must be made to insure a 50-year life for the lifting chains.

The lifting chain will at a minimum be subjected to rain, snow, etc. For projects that maintain an upper pool, the portion of the chain that connects to the gate, however, will be submerged in the river. This will subject the chain to silt and debris. Because the dam gates are rarely moved completely out of the water, the lower section of chain will be submerged for a majority of its service life. This lower portion of chain will also be subjected to sandblasting and paint over spray when the dam gates are being painted.

Recent lifting chain design utilizes aluminum bronze sidebars and stainless steel pins, see Figure 3-2. Both of these materials should provide adequate corrosion protection to allow the chain to last 50 years. Aluminum bronze is manufactured per ASTM B505 and a 62,000 psi (427,586 kPa) minimum yield strength is specified.

Figure 3-2. Typical Chain Installation

The stainless steel pins are manufactured per ASTM A564 Type XM-25 Condition H1050. This stainless steel is equivalent to Type 304 stainless steel for corrosion resistance. The primary disadvantage of using this type of stainless steel is that it was developed by one manufacturer and is not readily available from other manufacturers. Other options for the stainless steel include using ASTM A564 Type 630. This material is very close in properties to XM-25 and is available from more manufacturers. There are some disadvantages of using Type 630 stainless versus XM-25. The Type 630 stainless is more difficult to machine and must be age hardened prior to using. A comparison of the stainless steels is provided below in Table 3-1.

Table 3-1

	ASTM A A564 Type XM-25, H1050	ASTM A564 Type 630, H1025
Min Tensile:	145,000 psi	155,000 psi
Min Yield:	135,000 psi	145,000 psi
BHN:	321	331
BHN is Brinell hardness number		

3.1.5.5 Cost. The cost of the lifting chain will primarily be a function of the materials used. Although carbon steel materials will have the lowest initial cost, it is likely that the underwater portions of the chain will need replacement over 50 years. The stainless steel and aluminum bronze chain design will thus have a lower life cycle cost including maintenance costs. Also, the nickel plating of the pins approaches the cost of a stainless steel pin.

The cost for various combinations of materials has been estimated (per pound of chain). Cost estimating the chain on a per pound basis allows comparison of different designs. The chain length and size becomes irrelevant. All costs have been converted to 2001 dollars. The cost figures include all machining, assembling, and shipping. The following summarizes the chain costs among various supply contracts:

- All stainless steel chain using ASTM A564 Type XM-25 (both sidebars and pins) $10 to $11 per pound of chain.
- Steel Sidebars with Nickel Plated Pins and Grease Fittings - $3.3 - $4.8 per pound of chain
- Steel Sidebars and Nickel Plated Pins and Non-Lube Bushing - $8.7 per pound of chain
- Aluminum Bronze Sidebars and Stainless Pins - $6.5 per pound of chain

3.1.5.6 Chain Design. Chain design is based on a 50 year service life. Several design considerations need to be analyzed to insure this 50 year life. Strength and material selection are probably the most important. As discussed above, the material selection will dictate how much

the chain will corrode over 50 years (in particular the lower section of the chain). There are other design considerations that need to be analyzed, however. This includes yield strength, shear strength, fatigue strength, bearing stress at the chain joint, bearing clearance at the chain joint, and shock loading.

The ANSI/ASME standards define minimum ultimate strength (MUS) as the tensile load in pounds (or kilonewtons) at which a chain, in the condition at the time it left the factory, may break in a single load application. The yield strength of the chain should be 40% to 60% of the MUS. The chain also should be designed for shock loading. An example of this would be when a gate falls against a slack chain. The ACA Design and Applications Handbook lists a service factor of 1.4 to 1.7 for heavy shock loading. Several of the lock sites have broken chain in the past when a gate has been dropped against a slack chain or when slack chain was generated to provide additional momentum for breaking a frozen gate loose. Even though these practices are not recommended by the designers, the lifting chain will likely be subjected to these conditions over its service life.

Since the lifting speed of dam gates is very slow, the chain/sprocket design is not paramount. The main factor for the chain is the ability to hoist and hold the load from the dam gates and perform under all service conditions.

The interface between the pin and sidebar of the chain (or the chain joint) will be the highest stress area of the chain. A chain failure will result from either a sidebar or pin failure in this area. At the chain joint area, the sidebar will be in tension and shear. Corrosion in the chain joint area may cause the pin not to rotate as the chain is going over the sprocket causing damage to the gate hoist machinery. The bearing clearance necessary in the chain joint will depend on the materials used for the sidebar and pin. A minimum clearance of .005" is used in the current chain design. This value should be doubled or tripled if steel sidebars and pins are being used.

The pin undergoes bending stress in the center between sidebars and also shear stress at the chain joint. Both of these values need to be calculated.

An appropriate design standard is necessary to adequately design the chain joint area and determine a bearing stress. The American Association of State Highway & Transportation Officials (AASHTO) standard for bridges can be used for this purpose. In particular, the design constraints for pins, rollers, and rockers for bridges can be utilized. This standard makes a distinction between bearing stress on pins subject to rotation versus non-rotating pins. The chain joint should be classified as a rotating joint as opposed to a non-rotating joint. The standard also sets an allowable shear stress (Fv) of 40% of yield for the pin.

The AASHTO standard also helps determine whether a bushing or bearing is required. The AASHTO Specification for Highway Bridges, 15th Edition, 1992, Section 10 (Structural Steel), Part C (Allowable Stress Design), Table 10.32.1A permits an allowable bearing stress of 80% of yield for pins not subject to rotation. This specification allows a bearing stress of 40% of yield for pins subject to rotation. The standard states the effective bearing area of a pin shall be the diameter multiplied by the thickness of material on which it bears (the sidebar for instance).

The AASHTO standard for pins, rollers, and rockers is meant to eliminate galling in the pin/rocker area (ie. chain joint). The AASHTO standard implies that stress values below 40% of yield strength will avoid galling and that a bushing or bearing is not required. The standard only recognizes structural steel and alloy steel materials, however. Galling results from metal to metal contact. When a cohesive force between two metals exceeds the strength of either metal, adhesion or cold welding will occur. Under high stresses, cold welding will occur more rapidly and over a wider area. For instance, galling will likely occur when the chain is loaded up to and beyond yield limits. Galling is also a particular concern when stainless steels are mated with other stainless steels. Thus, if no bearing or bushing is used with an all stainless steel chain (sidebar and pin), the 40% of yield value may need to be lowered. The surface finish at the chain joint will also affect the rate of galling. The smoother the surface finish, the less likely galling will take place.

Chain designs using aluminum bronze sidebars and stainless steel pins will act like a bushing/pin interface. These two metals have good compatibility in terms of their bearing properties. These materials also have a fairly low corrosion potential (from dissimilar metal corrosion or galvanic corrosion). The lower the potential difference, the less likely galvanic corrosion will occur. The Metals Handbook, Volume 1, Properties and Selection of Metals, 8th Edition, American Society for Metals, lists a potential difference of +79 millivolts between aluminum bronze and 304 stainless steel in dilute sea water. This compares to +904 millivolts between zinc and copper.

Fatigue strength of the chain should be considered in the chain design even though the chain speed is slow. At many Corps projects, it should be noted, however, that fatigue strength is not likely to be the limiting factor in the chain design. This needs to be checked for each specific application. For Saint Paul District, it was assumed that the gates will be raised and lowered 3 times per week, then over a 50 year period, the chains will be cycled nearly 10,000 times. As each chain link section goes over the sprocket, it will be subjected to maximum tension. The link section will then be slack as it goes over the sprocket and is coiled up in the chain rack.

3.1.5.7 Maintainability. A primary goal of the chain design was to either eliminate or reduce the amount of maintenance necessary on the gate lifting chains. For projects that normally maintains an upper pool, a reasonable assumption can be made that it takes a crew of 4 people one week to bulkhead a single gate, temporarily support the dam gate, and grease the lifting chains (2 per tainter gate). Thus, switching to a non-lubricated chain offers a significant cost savings over 50 years.

When compared to replacing existing chain with wire rope, chain replacement offers several advantages. First, the existing gate lifting machinery could be reused. Also, using chain instead of wire rope requires less maintenance over 50 years. Wire rope needs to be lubricated on a regular basis. Any damaged part of chain can be replaced while wire rope must be completely replaced.

Many original (1930's design) lifting chains for the tainter gates were lubricated in a number of different ways. All of these lubrication methods allowed oil and grease into the water. Some of the lock sites lubricated the chain with 30W motor oil. Other sites used diesel fuel or waste oil. None of these methods allowed any lubricant into the chain joint since the bearing clearances were too tight.

Grease lubrication systems worked well initially, but there is a number of them now that will not accept grease. This system offered no advantages from a maintenance standpoint, and excess grease still ends up in the river.

The chain design presented herein use no bearing or bushing in the chain joint. The chain joint is designed as a bushing, however, since the sidebars are made of aluminum bronze and the pins are made of stainless steel. This design will eliminate the need for greasing of the chains.

3.1.5.8 Zebra Mussels. Zebra mussels have become more prevalent in the Upper Mississippi River system within the last several years. Zebra mussels attach themselves to submerged gates, intake valves, grating, concrete, etc. At a minimum, the submerged portion of the gate lifting chain needs to be designed to reduce or eliminate zebra mussel attachment. Material selection needs to be made to reduce or eliminate zebra mussels from attaching to the chain.

Testing and research by the U.S. Army Corps Of Engineers Waterways Experiment Station (WES) and the Construction Engineering Research Laboratory (USACERL) have shown that zinc and copper are toxic to zebra mussels.

The latest design of the chains, as stated above, use aluminum bronze sidebars and rollers. The specific alloy is UNS No. C95500 which is composed of 78% copper, and field inspections indicate little zebra mussel attachment to the lifting chains. Some zebra mussels were attached to the stainless steel collars and pins but no mussels were attached to the aluminum bronze sidebars.

3.1.6 General Design Considerations.

3.1.6.1 Hoist Arrangement. A hoist arrangement with one lifting point for each side of the gate is preferred over a single, center lift type hoist. This arrangement is essential for maintaining a minimum clearance between the lifting points on the gate when fully raised, and the service bridge structural steel. A typical hoist arrangement for the round link chain consists of a cross shaft, and centrally located motor/gear-box units with single reduction gearing at each end. This form of arrangement is generally considered to be the most economical from a viewpoint of space utilization and accessibility for maintenance. Because the gate is lifted from each end, it is possible that one lifting point could be subjected to a greater proportion of the maximum torque available in the drive train, due to becoming frozen-in or otherwise stuck. The motor design principally governs the maximum value of this stall or breakdown torque, which has an upper limit of 280 percent of the normal rated torque. A summary of manufacturers service factors for cataloged parts of a hoist arrangement is provided in Appendix C, for guidance purposes only.

3.1.6.2 Design Criteria. The maximum load on the hoist usually occurs at the beginning of

the hoisting cycle but it may occur at other points in the cycle, depending upon the position of the center of gravity of the gate with respect to the horizontal centerline of the trunnion. The total load at the hoist or hydraulic cylinder is the sum of the gate torques causing closure plus the gate torques resisting closure divided by the perpendicular distance from the centerline of the chain, rope or cylinder to the centerline of the gate trunnion. The gate torques causing closure are composed of:

- Weight of gate times the distance from trunnion to center of gravity.
- Weight of any silt or ice load times its effective moment arm.
- Loads due to fluctuating tail water from WES Model Test Data.

The gate torques resisting closure are composed of:

- Side seal friction (coefficient of friction of 0.5) times its effective moment arm.
- Trunnion friction (coefficient of friction of 0.3) times the radius of the trunnion bearing.

Computer programs aid in optimizing the capacity of either type hoist, with respect to hoist location, pier height and length, cylinder bore, pressure and stroke, including the magnitude and direction of the trunnion reaction. For the wire rope type hoist, the effect of the rope leaving the drum on the upstream or downstream side and its effect on bearing reactions can also be considered. Electrical equipment should be designed and selected in accordance with the provisions of UFGS 16905A. Tainter gate design loads are discussed in EM 1110-2-2702. Sample stress, sizing and power computations are provided in the Appendix C.

3.1.6.2.1 Hoisting Speed. A hoisting speed of about 5 mm per second (1 foot per minute) has been found satisfactory for most installations. However, the hoist speed should be varied so that the horsepower requirement will approximately match a standard motor rating. The designer should always discuss the gate raising and lowering speeds with the Hydraulics and Hydrology engineers.

3.1.6.2 Speed Reduction. For the main gear reduction, helical or herringbone type speed reducers are more efficient and generally less expensive than double reduction worm gear reducers, but if the latter type is used, ratios offering the best efficiency should be selected since the self-locking feature of units with small helix angles and low efficiencies serve no useful purpose on these hoists. Ratings of the reducers are based on the rated horsepower of the motor or on full load torque, depending on the type of reducer and the operating speed, reduced by suitable allowance for friction losses. Typical service factors for speed reducers are provided in Appendix C.

3.1.6.2.3 Brakes. The electric brake should be installed on the input shaft of the first speed reducer opposite the driving motor. This arrangement permits either the brake or motor to be serviced or replaced without disturbing the other.

3.2 Vertical Lift Gate (Spillways).

3.2.1 General. This sub-chapter presents two different types of operating systems for vertical lift gates, a hydraulic cylinder that is directly connected to the gate, and a screw stem hoist. In addition to these hoists, the hoist arrangement presented in Chapter 2 for the vertical lift gate for lock application can also be used for vertical lift gate for spillway application.

3.2.2 Direct Connected Hydraulic Cylinder.

3.2.2.1 General. Simple or telescoping hydraulic cylinders can be direct coupled to vertical lift gates. The choice depends on the extended and contracted lengths required for the application. The cylinder is hung from a stationary bracket and the gate is hung from the cylinder rod. The gate is raised when the cylinder is put in tension and it is lowered when the cylinder assembly is put in compression. Fluid can be supplied to the cylinder in a number of ways. The simplest system would provide one variable displacement type pump for each cylinder. With this type system the gate speed and position can be controlled by the pump's discharge or by valving. A constant displacement pump can be used for one or more cylinders in conjunction with an accumulator tank. With this type system the gate(s) speed and position must be controlled with valving. Backup power systems not generally recommended because of the high cost. Note however that an auxiliary lifting system such as a gantry crane must be provided for removal and/or replacement of the cylinders and their attachments. Note that it is assumed that each gate will be operated by one cylinder. If the installation requires a cylinder at each end of the gate, information in Paragraph 3.1 on cylinder synchronism for direct connected hydraulic cylinder gates should be reviewed. The choice of either one or two connection points depends on the gate height to width ratio and needs to coordinated with the gate designer.

3.2.2.2 Advantages. Direct connected hydraulic cylinder actuation offers several advantages compared to actuation by rope or chains. The most significant advantages are for closure during flow conditions. An hydraulic cylinder can push a gate closed and can do so at a controlled speed. Rollers are not needed for hydraulic cylinder actuated gates. Any tendency for a gate to stick will not have much effect on gate speed. A stick/slip situation will not arise. Hydraulic cylinder actuation will not allow a gate to jump under flow at the moment of final closure as may result if ropes or chains are used. Also, hydraulic cylinder actuation can provide better indication for gate travel. In addition to position, pressure which is proportional to force, can be monitored. This could indicate gate friction problems or if normal equipment capacity is being exceeded. Hydraulic cylinders and their required attachments are compact for fitting in new installations or retro-fitting in existing installations.

3.2.2.3 Disadvantages. The pumps and appurtenant equipment are fragile. However, they can often be located remotely, where they will be in a protected environment and will not be in the way of other equipment. Also, it is critical that the surfaces of the cylinder rods are protected from being bent, scratched, dented or nicked, and that they be of a material which is resistant to corrosion because of the environment in which they are located. The potential for a fluid leak into the waterway is always present. See Chapter 5 for design methods and materials that can minimize the disadvantages.

3.2.2.4 Design Criteria. The following information is from experience gained from powerhouse intake and draft tube gates and is being applied to a spillway structure. In addition to the weight of the gate and gate friction, design calculations for loads for hydraulic cylinder actuated gates must consider the force of gravity on the oil in the hydraulic lines and cylinder, the hydraulic friction of the oil passing through the piping, and the static and dynamic friction of the seals and scrapers on the cylinder rod. When hydraulic cylinder actuation is installed for the capability of lowering the gate during full flow condition, the design calculations should consider the downward force on the gate caused by water flowing under the gate at near closed positions.

3.2.2.2.1 Component Ratings. The hydraulic system components should be rated for at least 100 percent of the pressure of the system will see at normal maximum load, and the cylinder assembly components and attachments should be stressed to no more than 20 percent of yield strength.

3.2.2.2.2 Method of Attachment. Hydraulic cylinder assemblies should be attached to the gate and structure in such a manner as to allow free rotation of the assembly if not attached at both ends, and be designed to resist buckling without relying on the stiffness of the attachments to either the gate or the structure.

3.2.2.3 Control. The simplest control method would be for a system with one variable displacement type pump used for operation of one cylinder. The gate's speed and position could in theory be controlled by a variable discharge pump. However, in actuality valving would probably be required. The controls for the valve and pump would normally be through a solid state programmable controller.

3.2.2.3.1 Control With Constant Displacement Pumps. Controls would necessarily be more complicated where one or more constant displacement pumps are used for operation of one or more cylinders. For this type system, an accumulator tank is usually provided to reduce the cycling of the pump(s). Gate speed and position are usually controlled by valving, again through a solid state programmable controller. Additional controls are required to initiate pump starting and shutoff. It is generally best to award a design/build contract for the entire system, including the controls, to insure that the controls are suitable for the pumps, valves, and cylinders.

3.2.2.3.2 Location of Controls. Gate controls should be located in the control room and at the gate bay piers at the service bridge level. In the case of an unmanned project, closure should be from a manned location. It may be wise to add an interlock feature for gate opening. On such a system the gate opening could only be actuated from the cylinder support area. This arrangement could be used for system maintenance and repair.

3.2.2.3.3 Indication. Gate position indication for hydraulic cylinder actuated gates is normally provided by a selsyn type instrument which can be read on digital displays, locally, remotely, or both. Cylinder pressure should also be monitored. This is recommended as pressure is proportional to the force required to move the gate, and may indicate problems such as increased friction. Limit switches are usually provided to activate control valves at the end of the cylinder stroke. Pump shutoff or start-up is normally automatically actuated when the system pressure exceeds or drops below set pressure levels.

3.2.3 Screw Stem Hoist

3.2.3.1 General Description. Screw stem hoists typically used at Corps projects are for sluice gates, backflow control gates at large pumping stations and for vertical lift gates at low to moderate heads. The hoists are pedestal mounted, use an Acme-type stem and can be hand crank, electric motor or hydraulic cylinder operated. One hoist, located at the center of the gate, or tandem hoists can be used. Tandem hoists are used when the gate width to height ratio exceeds 4:1. For purposes of this document only tandem electric motor-operated hoists for vertical lift gates will be presented. The tandem hoist is arranged with a screw stem unit connected to each end of the gate and a centrally located dual output motor operated drive with gear reduction and torque limiting capability. The dual output reduction gear is connected to the two screw stem units through driving shafts. A typical arrangement is shown on Plate B-80

3.2.3.2 Design Considerations.

3.2.3.2.1 Hoist Sizing. The hoist size is determined by the operating thrust required to open the gate under full operating head. The operating thrust consists of water load, and the weight of the gate and stem. Specifically, thrust due to water load is determined by the head and the coefficient of rolling and sliding friction of the gate in its slot. Screw stem hoists and electric-motor operators are standard manufactured units. The design/selection guidance provided by manufacturers should be followed.

3.2.3.2. 2 Stem Sizing. The stem is sized to take into consideration corrosion, tension loads during gate raise and buckling which becomes critical during closing. The stem diameter is determined by the material used and the different vertical loads in the system along with the output of the floorstand and the unsupported length of the stem. The operating thrust of the gate does not directly control the size of the stem. The column effect of the unsupported length of the stem during closing is an important design consideration. The vertical loads include the weight of the gate and the frictional loads of the rollers and the rubber seals. The gate buoyancy is typically neglected in the calculations to keep them more conservative. Stainless steel is the most common material for stems. It is corrosion resistant and has higher strength than bronze. Operating stems can also be bronze or cold rolled steel. A stainless steel stem should be used in a corrosive environment.

3.2.3.2. 3 Stem Screw Protection. Stem screws have a metal pipe or a plastic tube to preclude water from infiltrating into the operating mechanism and to protect the stem screw as it protrudes above the operating stand when the gate is in the fully open position. However, when the stem screw travels downward closing the gate it does not have protection similar to the one aforementioned. An expandable stem boot is recommended to keep bird droppings and airborne debris from attaching to the grease-covered stem. The boot should be suitable for weather and UV exposure, and be resistant to the effects of the stem lubricant.

3.2.3.3 Safety Devices. Safety control devices for an electric motor-operated device include both limit and torque switches. These switches are located in the motor-operated drive unit. Limit switches will stop operation of the screw stem operating mechanism at the fully closed and fully open positions of the gate. The torque switch will stop operation of the hoist unit if an obstruction blocks the gate during closing. Stem screws move at a low speed creating a high torque. A defective limit switch or torque switch could cause serious damage to the operating platform. When the torque or limit switch malfunction during gate closure the uplift force created could be as high as the force generated by the stall torque of the motor. Operating platforms are not normally designed for this condition. The result could be an uplifted operating platform or a buckled stem. The platform should be designed to preclude damage in case of switch failure.

3.3 Wicket Gate.

3.3.1 General Description. A wicket gate can be used for many different applications. For this manual the application will be restricted to those used to create a dam. A Wicket Gate or Wicket, as shown in Plates B-81 and B-82, is a structural framed member which is connected to the sill of the dam and is raised or lowered by mechanical means. The general shape of a wicket is a flat sheet or skin plate with structural reinforcing. Curved wickets have also been designed but model studies have indicated lower pressure areas under the wickets can cause problems when operating the gates. Wickets are designed to be raised and set at a fixed angle inclined against flow. Factors which contribute to the angle of the wickets are the length of the wicket, the stability of the wicket, the head the wicket must restrain and other factors. A wicket can be made of may different structural components depending upon its application. A wicket gate dam is a dam which is used to create a pool by raising a series of wickets to restrict flow in a river. When the wickets are not required to sustain pool they can be lowered to the bottom of the river allowing navigation traffic to pass over the dam without damage to the gates. Three different types of wickets are described and detailed in this manual, each having its own set of advantages and disadvantages. As of this document's publishing date, the Corps has only used hydraulic cylinder operated wicket gates on the Olmstead prototype test facility. Test results were published in a paper presented at the 1998 Heartland Technology Transfer Conference, titled Results of the Olmstead Prototype Hydraulic Operated Wicket Dam.

- Manual Operated Wicket
- Retractable Hydraulic Cylinder Wicket
- Direct Connected Hydraulic Cylinder Wicket

3.3.1.1 Manual Operated Wicket. The manual operated wicket, Plate B-81, is comprised of five basic components: the base frame, wicket, horse frame, prop, and hurter. A sixth component is the piece of equipment which is used to raise or lower the wicket. Depending upon the size and location of the wicket dam, various mechanical operating equipment can be used to operate the wickets.

3.3.1.1.1 Raising. The basic principle of a manual operated wicket is a boat or some type of machine is maneuvered over the wicket and used to raise or lower the gate. The wicket uses a frame to assist in the lifting operation of the gate. The frame is commonly called a horse and is made of structural steel and connected to the sill on one end and to the mid-section of the wicket on the other. Traditionally the manual operated wickets have been raised by connecting the lifting mechanism on the upstream end of the wicket. The wicket is lifted off the sill from the upstream end and pivots about the horse connection. The horse rotates forward as the wicket raises. Current assists in the lifting operation by flowing under the gate as it raises and rotates about the horse frame. A prop is connected to the downstream side of the wicket and follows a track in the hurter located on the sill of the dam. When the horse has reached a designated angle with the sill the prop is designed to fall into a notch in the hurter. The lifting mechanism to the wicket is released and current holds the wicket at an incline. The wicket must be rotated about the horse connection to create the dam. Traditionally a device is used to push the upstream end of the wicket down until current overcomes the wickets center of gravity and forces it to come in contact with the sill. Once the wicket has rotated, its in its permanent raised position and the gate is set. The rotation point on the downstream side of the wicket is positioned so the combined forces below the rotation point is greater then those above the point, including those of ice loading. This prevents the wickets form flipping over unexpectedly.

3.3.1.1.2 Lowering. To lower a manual wicket the downstream connection point is used. The wicket is rotated forward against flow beyond a fixed angle which releases the prop from the notch in the hurter. Once the prop is clear of the notch, the wicket is allowed to fall by gravity to the sill. It is important to have some downstream pool to cushion the impact the falling wicket. The hurter is designed to realign the prop for the next lifting operation of the wicket.

3.3.1.2 Retractable Hydraulic Cylinder Wicket. The retractable hydraulic cylinder wicket, Plate B-82, uses the same principles as the manual wicket with the following modifications: A hydraulic power unit is used with hydraulic cylinders to operate the gate. The hydraulic power unit can be located on shore above flood stage or in a gallery beneath the wicket depending upon the size of the dam being built. The wicket is connected directly to the sill on the upstream end and does not use a horse frame.

3.3.1.2.1 Raising. The wicket is raised into position by rotating it from downstream to upstream about the hinge of the gate. The retractable design comprised of two cylinders, one for raising and lowering the wicket and one for aligning the raising and lowering cylinder. The hydraulic cylinders are mounted under the wicket. The lifting cylinders piston rod is mounted with a cup which engages a ball mounted on the downstream side of the wicket. The wicket is raised by extending the lifting cylinder which engages the ball and rotates the wicket to a fixed angle where a prop engages a notch in the hurter in the same manor as the manual operated wicket design. Once the prop is set in the notch, the combination of current and gravity of the inclined wicket keep the prop securely fixed in the hurter. The piston rod is retracted to remove it from potential damage caused by debris.

3.3.1.2.2 Lowering. To lower the wicket a second smaller alignment cylinder is used to align the larger lifting cylinder to the proper angle to contact the cup with the ball and the gate. The piston rod rotates the wicket forward until the prop clears the notch in the hurter and the flow of fluid out of the cylinder controls the speed at which the wicket lowers.

3.3.1.3 Direct Connected Hydraulic Cylinder Wicket. The direct connected cylinder design, Plate B-82, is very basic. One cylinder is directly connected to the backside of the wicket. The connection is made at the same location the prop is connected to in the retractable cylinder design. To raise the gate the cylinder is pressurized and the piston rod extends rotating the wicket to 65-degree. The hydraulic valves hold the wicket in the raised position until the wicket is lowered.

3.3.2 Design Criteria.

3.3.2.1 Wickets. Wicket gates are considered a type A hydraulic steel structure per EM 1110-2-2105. Design loadings were applied for allowable stress design in accordance with EM 1110-2-2105.

3.3.2.2 Mechanical Components. Pins, tie downs, etc., are to be designed for the maximum loads or forces applies to them with a minimum safety factor of five based on the ultimate tensile strength of the material involved. In addition, each component should be designed for a unit stress not to exceed 75 percent of the yield strength of the material.

3.3.2.3 Bearings. All bearings associated with a wicket dam are to be self lubricating type and are to be designed not to exceed a maximum load of 55 Mpa (8,000 psi)or the manufacturers recommendations. Bearings should have a finish of 16 micro inch or better. A minimum of 1:1 ratio of length to width should be used. Rotating Pins which operate with bearings should have a finish of 16 micro inch or better and should be made of stainless steel. Formula for determining bearing size:

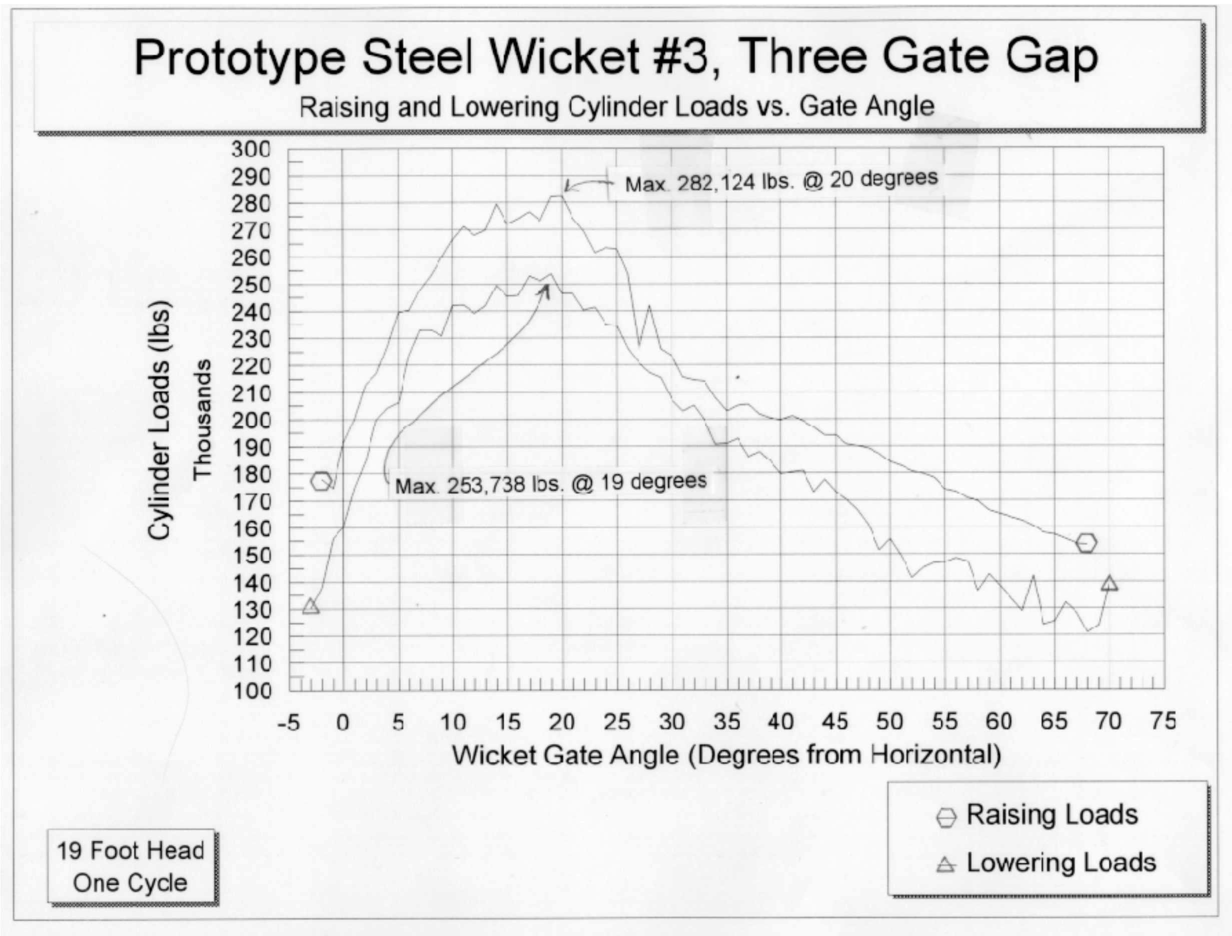
$$\text{Maximum Load} = \frac{\text{Maximum Applied Load}}{\text{(Shaft Dia.)(length of Bearing)}}$$

3.3.2.4 Hydraulic Cylinder and Piston Rod. The hydraulic cylinder is to be designed for extreme operating conditions, loads which operate up to the maximum drought condition. The cylinders are to be designed to operate the wicket against a ice load of 7,500 kg/meter (5000 lbs/linear foot) applied at the top of the wicket. Piston rods are to be sized with a factor of safety of at least 2.5 on the maximum load imposed on the rod and the critical buckling load. Critical buckling loads should be designed using Johnson or Euler equations. Piston rods should have a finish of 8 micro inch or better.

3.3.2.5 Model Test. Testing results and design loads which have been collected at WES and the Olmsted Prototype Hydraulic Operated Wicket Test Facility indicate the following.

3.3.2.5.1 Maximum Load. When raising a wicket the maximum lifting load on the components occurs with three wickets down and raising the middle wicket. The cylinder raising and lowering loads which have been collected at the Olmsted Prototype test facility for a steel wicket with the dimensions 2.8 meters wide by 7.8 meters long (2.2 feet wide by 25.5 feet long) are shown in Table 3-2.

Table 3-2



3.3.2.5.2 Hurter Design. When designing a wicket dam which uses a hurter and prop, design the hurter so the side return guide is located on the side of the hurter which is the normal direction which will be used when lowering the dam. Example if the wicket dam is designed to be lowered from right to left looking downstream then the hurter return rail should be located on the left looking down stream. The reason for this is to prevent current from forcing the prop to the side rail and not allowing it to transition through the hurter.

3.3.3 Controls.

3.3.3.1 Manual Operation. The manual operated wicket is controlled by the machine which is used to hook and raise or lower the gate. To prevent damage to the components the machine should not be allowed the raise a wicket in more than 10 seconds.

3.3.3.2 Retractable Hydraulic Cylinder. The retractable hydraulic cylinder control system should be designed to have a maximum raising time of a wicket of approximately 12 minutes. The maximum time to retract or extend the lifting cylinder piston rod under no load should be 3 minutes. The maximum lowering time of a wicket should be 3 minutes. The minimum time to extend or retract the alignment cylinder piston rod wall be 10 seconds. A series of wickets are to be operated off one power unit.

3.3.3.3 Direct Connected Hydraulic Cylinder. The direct connected hydraulic cylinder control system should be designed to have a maximum raising time of a wicket of approximately 12 minutes. The maximum lowering time of a wicket should be 3 minutes.

3.4 Hinged Crest Gate.

3.4.1 General Description. Hinged crest gates are mounted (hinged) at the crest (invert) of the spillway. There are many ways to operate a hinged crest gate but they all lower to open and raise to close. This manual will provide information about the torque tube and hydraulic cylinder type of support and operating system. Details of this type of gate and operating system, used at the Montgomery Point Lock and Dam, is provided in Plates B-83 and B-84. Hinged crest gates, similar to wicket gates, require no intermediate piers and therefore provide no physical restriction to river navigation when the gates are in the down position. Also when lowered, the gates can be designed to conform to the shape of the spillway. Other unique features include the absence of operating machinery exposed to river flow and debris damage and overtopping flows can occur with minimal vibration.

3.4.2 Design Considerations. The height and width of the gates are restricted because of the size of the torque tube required for larger gates. The hoisting mechanism, in addition to raising and lowering the gate, must be able to hold the gate at the desired raised position. A dogging device should be provided to hold the gate in the lowered position during major maintenance, repair or replacement of the operating machinery. The gate operating machinery should be designed to withstand the normal 3-dimensional loading imparted from the gate into the cylinder/cardan frame assembly as well as the much larger barge impact loading. A separate pressure relief system should be incorporated in the hydraulic system to protect the gate and machinery from barge impacts. Speed and ease of maintenance are primary considerations for the gate assembly. A barge mounted crane must be used to remove or replace the gate which is the largest and heaviest member of the system.

3.4.3 System Components. The main components of a hydraulic cylinder operated torque tube gate are the support structure, cardan assembly, hydraulic cylinder assembly, operating arm assembly, gate drive shaft, and the torque tube gate.

3.4.3.1 Support Structure. The support structure includes the support for the gate and the support for the hydraulic cylinder assembly. The torque tube gate assembly is supported on bearings at intervals along its length. The bearings are either self-lubricating or grease lubricated, however self-lubricated is preferred. Information about self-lubricated bearings is provided in CERL Technical Report 99/104, Greaseless Bushings for Hydropower Applications: Program, Testing, and Results. The bearings are supported in saddles that are anchored to embedded structural elements. The hydraulic cylinder is supported by the cardan assembly. The cardan assembly is supported by a fabricated steel weldment that is bolted to the top of the embedded section to facilitate installation and removal of the cylinder.

3.4.3.2 Cardan Assembly. The cardan assembly provides support and minimize eccentric loading by allowing the cylinder to freely rotate in any direction as required for gate operation. The cylinder trunnion is mounted vertically in the cardan ring to allow the cylinder to rotate horizontally from side to side. The cardan ring is pinned horizontally to the support structure to allow the cylinder to rotate vertically. The bearings of the cardan assembly are self lubricating bronze with thrust flanges. The cardan frame is constructed from steel plate.

3.4.3.3 Hydraulic Cylinder Assembly. The main components of the hydraulic cylinder assembly are the cylinder tube, piston rod and trunnion. The cylinder tube is normally fabricated from one piece AISI 4340 steel and is fitted with a one piece ASTM A36 steel trunnion. The trunnion pins are fabricated with stainless steel wearing surfaces to bear on self-lubricating bushings. An ASTM A668, Type NH steel cylinder rod end is fabricated with a spherical joint rod end for connection to the gate operating arm. The spherical joint is furnished with a self-lubricating bushing. It's important to include a 38 mm (1.5 in.) reserve stroke to the working stroke of the hydraulic cylinder for erection tolerances and other factors like bearing wear. The trunnion is located near the center of the cylinder. The gate operating system is located in an equipment gallery. It's important when designing the gallery to provide an access shaft to allow the cylinder to be lowered into the gallery from the land wall side. The equipment is moved in the gallery on wheeled carts.

3.4.3.4 Operating Arm Assembly. The operating arm assembly connects the hydraulic cylinder to the one piece drive shaft. The assembly is fabricated from two forged steel arms one right hand and the other left hand. The hub end of the arms is split and bolted together around the drive shaft. The hub is permanently welded to the drive shaft after all field adjustments have been made. To increase the strength at this joint, inner and outer rings are welded to both the operating arms and the drive shaft. The other end of the arms form a clevis and connect to the rod end of the operating cylinder via a single corrosion resistant pin. Each assembly is equipped with a dogging device. The dogging device consist of a 90-degree rotating shaft, that is pinned on one end and threaded on the other. The pinned end is connected to a bracket that is anchored to the concrete alcove wall. The threaded end is swung into the dogging slot of the gate operating arm and tightened up. The dogging device will be held in an inoperative position by a latch back pin when not in use.

3.4.3.5 Gate Drive Shaft. Both ends of the drive shaft extend through the alcove cover and connect to the torque tube. The drive shaft is supported by two self lubricating bearings at both alcove penetrations. Leakage past the bearings is prevented by two split rings seals on the waterside and by four layers of split chevron packing on the inside. The middle of the drive shaft connects to the operating arm assembly. The drive shaft is fabricated from a one piece hollow steel forging. Both ends of the shaft are machined with a tapered slot for a wedged connection assembly with the torque tube to ensure a tight fit and ease of assembly and disassembly.

3.4.3.6 Torque Tube Gate. The torque tube gate is supported by the torque tube bearings one on each side of the alcove cover. The two torque tube portions of each crest gate are fabricated from rolled carbon steel plate. The drive flanges of the gate for connection to the gate drive shaft are machined from steel forgings and are welded to the torque tubes (key vertical when the gate is in the fully lowered position). The free ends of the torque tube are closed watertight with welded steel plate. Stainless steel bearing sleeves are provided at bearing locations.

3.4.3.6.1 Bearings. The torque tube bearings are of the pillow block type. The housing is a horizontal split steel casting with self lubricating bearing with seals to prevent entrance of sand and silt.

3.4.3.6.2 Thrust Loading. The side ways or thrust loading of the hinged crest gate is restrained by a 155-degree stainless steel wear plate anchored to the bottom half of each torque tube bearing housing. The corresponding mating surface is three ultra-high molecular weight polyethylene (UHMWPE) pads mounted on the torque tube.

3.4.3.6.3 Access. The alcove cover provides access for installation and maintenance of the gate operating machinery. It consist of a removable steel top fabricated from a steel weldment, two removable side wall supports fabricated from the steel upper bearing casting with associated weldments, and an embedded portion fabricated from a steel lower bearing casting with associated weldments. The castings provide the bearing support for the drive shaft bearing. Neoprene seals and o-rings are provided at all mating surfaces to prevent leakage.

3.4.4 Gate Control.

3.4.4.1 Operator Stations. The dam gates can be operated from several different operator stations. These systems are normally operated remotely from a personal computer located at the operator's station in a control tower. A redundant local system is also provided in the dam gallery. If all of the electronics fail, manual control is provided near each hydraulic cylinder for individual operation of the gate, when needed.

3.4.4.2 Hydraulic Power. The gate cylinders for multiple gates should be supplied with hydraulic power from at least two main hydraulic power units (HPU) as well as from two separate accumulator HPU's. All HPU's are typically located in a control tower. The main hydraulic system raises and lowers the dam gates one at a time. A typical operating hydraulic pressure is 13.8 Mpa (2,000 psi). The accumulator system holds the gates in either the fully raised or fully lowered position through separate supply piping to either the bore end or the rod end of the cylinder. The accumulator system is sized such that the accumulator HPU will charge the accumulator approximately once every two weeks due to expected fluid leakage in the hydraulic components.

3.4.5 Gate Operation.

3.4.5.1 Normal. During normal operation, the safety measures have no effect on the gate raise and lower cycle.

3.4.5.2 Gate Drift. For gate drift trouble, an indication signal light and audio alarm should be provided to notify operator of most accumulator or leakage problems. Activation of the main HPU should temporally remedy the situation until the lock personnel can fix the problem.

3.4.5.3 Debris Overload. A signal light and audio alarm should be provided to notify operator of debris build-up. After lowering the gate to flush the debris the hydraulic system should return to normal operation.

3.4.5.4 Impact Release System. A gate impact release system should be provided to prevent damage to the hinged crest gate in the event of an impact from a runaway barge, some other large object or debris loading. The system requires the rapid relief of hydraulic fluid from the cylinder full-bore end to the rod end and an auxiliary reservoir. The auxiliary reservoir has the same capacity as the piston rod volume. The system utilizes only hydraulic pressure for release and no electrical power or signals are required for operation or activation.

3.4.5.4.1 High Pressures. For high pressures the impact release system installed at the Montgomery Point Lock and Dam project uses two logic control valves (one redundant) mounted directly on the full-bore end of the hydraulic cylinder to regulate the rate of hydraulic fluid flow out of the cylinder. These valves open by hydraulic pilot lines when unusually high pressure is generated in the full-bore end of the cylinder by impact. The directional control valve, which supplies pilot pressure for the actuation of the logic control valves, is equipped with a detent to ensure that the directional and logic control valves remain open and the gate continues to lower after the pressure in the cylinder falls below the value generated by the impact. The directional control valve is initially actuated by hydraulic pilot pressure released by relief valves (two relief valves should be provided, one redundant). The directional control valve is provided with an electronic solenoid to reset the valve (the solenoid plays no part in the release) for normal operation of the gate. The reset will be performed by a push-button mounted at the gate local control panel. Once the gate reaches its neutral position (the gate position where the net torque on the gate is zero), the gate will be lowered to the fully lowered position by starting the hydraulic system pump.

3.4.5.4.2 Extreme High Pressures. For even higher pressures spikes the impact release system could be provided with additional pressure relief valves set at a slightly higher pressure than the impact release trip pressure to protect the cylinder and hydraulic system from extreme high pressure transients (spikes) that may occur during impact. As soon as the pressure fall below the set point these relief valves stop relieving.

3.4.5.4.3 Reset. After an impact release, the electrical solenoid on directional control valve is manually energized from a push-button on the gate control panel to shift the directional and logic control valves back to their normal positions. After impact release the hydraulic fluid in the auxiliary reservoir must be drained into a suitable container and manually returned to the HPU reservoir located in the control tower.

3.5 Submergible Tainter Gate.

3.5.1 General. The machinery used to operate submergible tainter gates usually consists of two equal hoist units of opposite hand design arranged to lift each end of the gate. The hoist units are kept in synchronism by power selsyn motors. Each hoist unit consists of a rope drum, open gear set, speed reducer, magnetic brake, hoist motor, and power selsyn. The drum is mounted on a cantilevered shaft of a size to prevent excessive error in the mesh of the final drive pinion and gear due to shaft deflection. A general arrangement of an electric-motor driven hoist for the tainter gate is shown in Plate B-85.

3.5.2 Design considerations and criteria. The design capacity of the hoist should be based on the maximum load at normal speed which is found to be at the nearly closed or raised position. The hoisting speed should be selected so as to raise the gate from full open to closed in 2 to 3 minutes, varying so as to allow the selection of a motor of standard horsepower and speed. General criteria applicable to the design and selection of various hoist components are presented in Paragraph 5.1. Shock, impact, and wear factors are considered negligible and may be disregarded. Selection criteria for wire rope is contained in EM 1110-2-3200. Drum diameter should not be less than 30 times the rope diameter.

3.5.3 Determination of machinery loads. The maximum dynamic load on the hoist normally occurs near the end of the raising cycle. The maximum holding rope load occurs when the gate is fully raised and the water level is below the upper sill. Detailed information on load requirements is provided later in this chapter and in EM 1110-2-2702. No consideration should be given to rope loads created by the flow of water over a partially opened gate. The rope loads from these conditions are indeterminate and control features are provided to prevent their occurrence. The total load on the rope drum is the sum of the following:

- Deadweight of the gate as applied to a moment arm (W x CG) divided by the perpendicular distance of the rope to the gate trunnion center line.
- Side seal friction (total seal force x 0.05).
- Weight of the ropes can be neglected.
- Trunnion friction.
- The static load of the water head on the unbalanced area on the bottom seal.

- Ice buildup and silt formation should be considered when severe freezing or siltloaded water are factors. Seal heating systems usually minimize these factors.